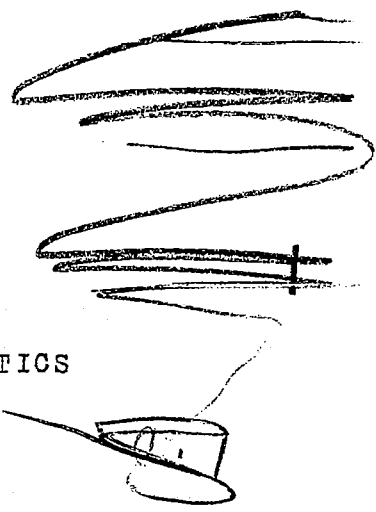TECHNICAL MEMORANDUMS

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

No. 875

HEAT-STRESSED STRUCTURAL COMPONENTS

IN COMBUSTION-ENGINE DESIGN

By Otto Kraemer

Washington
September 1938

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL MEMORANDUM NO. 875

HEAT-STRESSED STRUCTURAL COMPONENTS

IN COMBUSTION-ENGINE DESIGN*

By Otto Kraemer

Heated structural parts alter their shape. Anything
which hinders free heat expansion will give rise to heat
stresses. Design rules are thus obtained for the heated
walls themselves as well as for the adjoining parts. An
important guiding principle is that of designing the heat-
conducting walls as thin as possible.

Machine-structure materials expand on heating with a
resulting change of shape of the structural part. Partic-
ularly, for those structural components that operate with
a small amount of clearance, this fact must be taken into
account and leads to known design rules: namely, increased
clearance of the cold piston, roller play in valve-gear
drive, axial play in crankshaft bearings, maximum possible
symmetry of pistons, bearing sleeves, etc.

Any prevention or restriction of the heat expansion
is associated with extremely large forces. The recognition
of this fact has led to the fundamental principle, namely,
that of allowing the heat expansion to take place freely
and unobstructed as, for example, in the case of the liner
that is fixed at one end, or the axially free piston pin,
and similar examples.

Where it is not possible for a free relative displace-
ment of the differently expanded parts to take place, free
yielding of one structural part must be provided for in
order that the deformation produced by the adjoining struc-
tural part may be possible without unduly large forces and
stresses. This is most clearly realized in the case of ex-
haust pipes whose bellows-shaped intermediate sections
make the entire structural part freely yielding. In the
case also of all screws and heat-deformed structural parts,
the principle of making the screws extensible and capable
of bending was recognized – that is, long and thin, in or-
der that changes in shape could take place without too great

additional stresses. The stress due to the heating may
therefore in all cases be avoided or greatly restricted.
Only a hindering of the free expansion gives rise to the
stresses.

The requirement of unhindered deformation is, however,
difficult and almost impossible to satisfy in the case of
those structural components through which the heat must
flow. As heat flow is caused by temperature differences,
these differences occur in one and the same structural
part and may be different at various sections. In a struc-
tural component, however, in which hot and cold portions
are adjacent to each other, considerable heat stresses
must arise if the expansion of the hot portions is prevent-
ed by the nonexpanding cold portions.

It is of course impossible to avoid the conduction of
the heat through the walls of the engine. The ideal ther-
modynamic processes would require heat-insulated walls
since the removal of heat from the working cycle impairs
the thermal efficiency and thereby raises the fuel con-
sumption. The walls of the combustion chamber must be
cooled mainly, however, in order to preserve them from de-
terioration. Reciprocating engines, and this is their
main advantage over combustion turbines, do not have any
portion of their walls continuously exposed to the maximum
temperature and are automatically kept relatively cool by
the rapid interchange of the hot and cold gases. The aver-
age temperature, however, which the walls assume, is much
too high for artificial cooling to be dispensed with. The
materials of the cylinder head, piston, and liner would
become soft at these temperatures which are of the order
of magnitude of 700° and wear off and, in particular, would
become entirely useless as lubricated contact surfaces.

APPLICATION OF HEAT RESISTING PROTECTIVE PARTS

A noncorrosive material of low conductivity would be
most desirable for cylinder head, piston, and liner. Such
an engine, which on account of its hot walls could work
only on the Diesel cycle, would conduct away only small
heat losses and therefore would suffer only slightly from
the heat stresses. The familiar special materials which
are applied for prechamber inserts and similar parts, for
which the automatic and artificial cooling described above
would be insufficient, are resistant at high temperatures

but their mechanical strength properties, as in the case
of usual materials, are much impaired (fig. 1). They may
not be applied directly as a mechanically stressed wall
but as a heat-protective cover for the cooler wall.

Figure 2 shows the piston of an engine with a heat-
resisting, mushroom-type piston covering. An air slit
surrounds the entire mushroom covering so that the heat
expansion of the hot "mushroom" takes place unobstructed
while the heat flow to the piston itself is very much re-
stricted.

Protective sheets for exposed cylinder-cover areas
have also often been applied for the same purpose. It was
found difficult, however, to connect these protective
sheets in such a manner as to permit the expansion of the
hot sheet against the protected cool cylinder cover to
take place without tearing or warping. This fundamental
difficulty will be encountered in all attempts at placing
a protective lining inside of the combustion chamber. The
protective material must withstand great heat and have the
least possible heat-expansion coefficient.

INTERNAL COOLING

Where the above means are not applied, it is neces-
sary to cool the walls. With respect to the stress of
the walls arising from the temperature differences in the
individual wall portions, it makes a great difference
whether the cooling is from within or without. The recip-
rocating engine with its periodically renewed charge of-
fers the possibility, as mentioned above, of advantageous
internal cooling. In the case of the two-stroke-cycle en-
gine, and also the four-stroke-cycle engine with charging
blowers, effective cooling of the walls may be attained
by the simple cooling of the inner walls with the cold
scavenging air. Engines with superchargers actually have,
in general, lower wall temperatures as a result of the in-
creased cooling effect of the fresh scavenging air in
spite of the higher power per unit displacement volume;
also, the temperature differences within the walls them-
selves are smaller (reference 1). The heat of vaporiza-
tion of the liquid fuel also has an internal cooling effect.

EXTERNAL COOLING

The case is different in the application of the usual
external cooling of the walls of the combustion chamber.
In this case the heat is conducted through the wall. The
cooled external side of the wall is considerably colder
than the inner side in contact with the gases, so that
large heat deformations or heat stresses must be set up.

Figure 3 shows the temperature variation at a straight
wall. The temperature difference $(\vartheta_1 - \vartheta_2)$ of the two
wall sides is directly proportional to the heat quantity
Q flowing through unit area per unit time. Although the
gas temperature at the heated inner surface fluctuates be-
tween very wide limits - from 50° to about $2,000^\circ$ - the
heat Q conducted through the wall is practically uniform.
Only the inner hot surface follows the rapid temperature
fluctuation but only to a very small extent - of the order
of $\pm 10^\circ$ to 20° (reference 2). The heat Q which prin-
cipally determines the temperature difference in the wall,
depends only to a very slight extent on the heat-conduc-
tion coefficient λ and wall thickness δ, provided one
is not dealing with heat-protecting materials such as may
be represented by scale, oil film, or air gaps. The value
of Q is primarily determined by the temperature differ-
ence between the gas and the wall and the heat transfer
coefficient from gas to wall. Thus it may be seen that
the wall will be most subject to the danger from heat
stresses where the combustion temperatures are highest
(small excess air quantity) or the walls are very cold
(sharp cooling) and where high velocities and strong turbu-
lence occur during the combustion.

It may further be seen from the simple relation
$(\vartheta_1 - \vartheta_2) = Q \frac{\delta}{\lambda}$, that the temperature difference $(\vartheta_1 - \vartheta_2)$
is directly proportional to the wall thickness, so that
thick walls have large temperature differences whereas
thin walls have small temperature differences. Other con-
ditions remaining the same, thin walls are therefore less
stressed by heat than thick walls, so that the fundamental
principle arrived at is, namely, that the walls should be
made only as thick as appears necessary from considerations
of the combustion pressures.

THE FLAT DISK AS A STRUCTURAL ELEMENT

It is possible to seek for the walls such shapes as will deform with the least possible restraint corresponding to the temperature distribution. This condition is satisfied to a high degree by the flat disk which when heated on one side deforms into a shape similar to that of a spherical shell with the concave side toward the heat source. A cylinder cover is therefore concave toward the piston, and a piston head toward the cover — the deformation being greater the less it is hindered by neighboring structural parts such as ribs, edges, tubular openings for valves, and passages. The wall thickness in the case of unhindered deformation plays only a small part.

Unfortunately, restrictions to the expansion are always present in the form of the adjacent structural parts, and in any case the flat disk is the most unfavorable structural shape with regard to its ability to withstand fluctuating pressures. In addition to the restriction against deformation and the stresses by the explosion pressures, there is the further disadvantageous circumstance that the more dangerous, by far, tensile stresses occur at the cold side in contact with the cooling water.

In order that these stresses remain small, either the neighboring parts hindering the deformation or the heated disk itself must be designed to be yielding, while the other part must be strong so as to be able to take up the explosion forces.

STIFF CYLINDER COVER WITH YIELDING NEIGHBORING

STRUCTURAL PARTS

Figures 4, 5, and 6 show stiff cylinder covers with yielding neighboring parts. On heating the covers a concavity is formed toward the hot chamber, which deformation may without effort be imparted to the thin-walled bellows-shaped neighboring parts. Figure 7 shows the valve opening in the cylinder cover completely free toward the outside. The cooling water space is made watertight by yielding, soft packing. In the piston (fig. 8) the ribs that would have prevented the deformation of the thick, hot piston head and would have stressed it to cracking, have been dispensed with.

The necessary stiffness of the cylinder structures required by the stressing of the fluctuating explosion pressures may be attained not only through large wall thickness but also by curving the surfaces and by the use of radial ribs. A curved surface is very stiff even for a thin wall and "breathes" much less under the alternating combustion pressures than a flat disk. Stiffening ribs on the cold side give great stiffness of cover against the explosion pressures and prevent the generally overestimated "breathing." On the other hand, these well-cooled ribs hinder the heat expansion of the hot exposed surfaces and give rise to large and dangerous stresses. The latter are greater the steeper the temperature change from wall to rib and the less yielding the rib and wall are designed (fig. 9). These ribs therefore are avoided in most cases on the basis of unfavorable experiences. Where ribs are applied a uniform temperature drop along the rib should be striven for by having a broad transition at the foot of the rib (fig. 10), for under such conditions the ribs follow most freely the deformation of the exposed walls. It must be remembered, however, that a hindering of the deformation of the walls as, for example, by a tubular valve passage in the cylinder cover, also means a hindering of the rib deformation and leads to a dangerous "strength contest" between ribs and valve passage.

The temperature jump between wall and ribs is particularly marked if the heat conduction is disturbed by scale formation, lime deposit, or oil film. The disk thereby assumes a greatly increased temperature and heat expansion, while the ribs immersed entirely in the water remain cold and are in danger of tearing the more the disk wall is restricted from expanding; that is, the stiffer and thicker it is. Such local irregularities of the heating and heat conduction may raise the heat stresses in an unpredictable manner. A very stiff cylinder cover such as the cone-shaped one shown in figure 11, that has a tendency particularly toward a nonuniform temperature because of the possibility of a lime deposit, tends to force the vertex of the cone downward in expanding and stresses the tubular valve passage in the stiff cover to the breaking point.

YIELDING COVER WALL WITH STIFF SUPPORTING STRUCTURE

The above considerations and experiences lead to the second possibility in design, namely, that of a yielding

exposed surface with stiff support. A yielding exposed
wall may permit a restriction on its heat deformation —
whether due to uniform or nonuniform or disturbed heat
conduction — without thereby giving rise to large forces
and stresses. Figures 12 and 16 show constructions with
relatively thin exposed plates and very strong supporting
structure.

If we go still further so that the thin exposed wall
is no longer capable of supporting the explosion pressures,
it is necessary to use supporting ribs for the same pur-
pose as stiffening bolts are used in boiler construction.
The supporting ribs could be designed as in figures 13 and
14 — that is, long and flexible. In this form they will
fulfill their purpose without giving rise to strong stress-
es in the thin wall during their heat expansion.*

SEPARATION OF THE HOT FROM THE COLD STRUCTURAL PART

In many cases it is attempted to design the hot and
cold parts as separate components. Since the hot part,
however, tends to deform by the heat expansion the diffi-
culty in keeping the two parts in proper relation to each
other is greater the thicker, that is, the less yielding,
the hot part is designed. If the pieces connecting the
parts, however, are so designed as to permit yielding to
the heat deformation, the stiffening effect of the cold
upper portion of the cylinder cover is removed and the hot
portion is able to "breathe" in correspondence with the
explosion pressures.

If the hot portion of the cylinder cover is in the
form of a freely suspended added piece, no "breathing" or
varying stresses from the gas pressure can take place.
The suspended piece (fig. 15), in whose place there may be
imagined a cooling pipe coil with forced water circulation,
serves as a protective device for the stiff upper cover.
In this case we have a heat screen not, as in the case
discussed above, of hot, noncorrosive material, but a
"cold" heat screen. On account of the undesirable heat
conduction the cold screen is naturally economically in-
ferior to the hot screen. Figure 16 shows, for compari-

---

*In connection with a thick wall, such supporting ribs
would naturally not only become useless but also unsafe
since they could scarcely hold out against the deformation
of a thick, hot wall.

son with figure 15, a recent cylinder-cover design of the same manufacture without freely suspended piece inside.

STRESSES IN A CYLINDER LINER

In the cylinder liner, which forms a part of the hot combustion-chamber walls, the deviation with respect to stiffness from the flat disk shape is greatest. For our brief consideration, a liner may be chosen whose wall thickness is sufficiently small in comparison with its diameter, and for simplicity it may be assumed that on heating, the same geometrical shape is maintained.

The hot inner layer is prevented from fully expanding and the cold outer parts are forcibly lengthened. Heat stresses are set up which are compressive on the hot side and tensile on the cold. With the notation of figure 3, there is obtained for the heat stress:

$$\sigma = \frac{\vartheta_1 - \vartheta_2}{2} \, \beta \, E = Q \, \frac{\delta}{2} \, \frac{\beta \, E}{\lambda} \quad (kg/cm^2)$$

where β is the linear heat-expansion coefficient, and E the elasticity coefficient.

With a value of $Q = 100,000$ k cal/m^2h, there is obtained, for example, for cast iron:

$$\sigma = 100000 \, \frac{\delta(m)}{2} \, \frac{0.00001 \times 1000000}{50} = 100 \, \delta(cm)$$

and for aluminun

$$\sigma = 100000 \, \frac{\delta(m)}{2} \, \frac{0.000024 \times 700000}{170} = 50 \, \delta(cm)$$

It may be seen that for equal heat transmitted, the heat stresses in the case of aluminum are exactly half as great as for cast iron, due to the good heat conductivity of aluminum which results in a small temperature difference in the wall. The effect of the thickness is of primary importance, however, and it is seen that thin liners (fig. 17) are least exposed to heat stresses.

Upon this constant heat stress there is superposed

the fluctuating stress due to the explosion pressures, so that the maximum stress of the cold outer fibers is given by the following expression:

$$\sigma_{max} = \frac{Q\beta E}{2\lambda}\,\delta + \frac{pD}{2}\,\frac{l}{\delta}$$

where D denotes the inner diameter of the liner.

In figure 18 this value has been plotted for various wall thicknesses. It is thus possible to use that wall thickness which gives the minimum value for σ_{max}. Comparing the curves of figures 19 and 20, in which the fluctuating stresses have been plotted against the mean stress, with the admissible stresses of the material, it may be seen that an increase in the wall thickness beyond that corresponding to the minimum value of σ_{max} brings no added safety.

Any obstruction in the heat conduction, for example, as a result of scale deposit, oil, air pockets, raises the total temperature of the liner but lowers the temperature drop between the inner and outer fibers. In such a case the wall will burn first from the inside before it will tear on the outside. A sharp temperature drop in the axial direction of the liner will naturally also give rise to stresses, and these shear and bending stresses will be larger the stiffer the liner - i.e., the thicker its wall.

Translation by S. Reiss,
National Advisory Committee
for Aeronautics.

REFERENCES

1. Pflaum, W. in 74 Hauptvers. d. VDI, Berlin, 1936, p. 252.

2. Pfriem, H.: Z. VDI, Bd. 81 (1937), p. 1477, particularly figure 12.
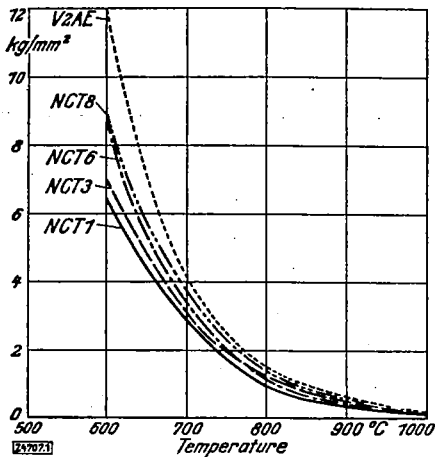
Figure 1.- Allowable continuous stress of various heat resisting alloys.

Figure 2.- Working piston of a Diesel engine with mushroom shaped piece on piston head.

$$Q = a_1 (t_1 - \vartheta_1) = \frac{\lambda}{\delta} (\vartheta_1 - \vartheta_2)$$
$$= a_2 (\vartheta_2 - t_2) = k (t_1 - t_2) \quad \text{kcal/m}^2 \text{h};$$
$$1/k = 1/a_1 + \delta/\lambda + 1/a_2; \qquad (\vartheta_1 - \vartheta_2) = Q \, \delta/\lambda .$$
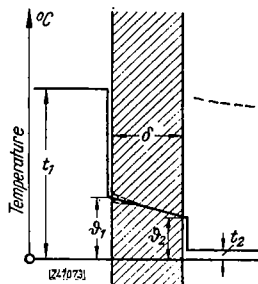
Figure 3.- Variation of temperature at a heat conducting wall

t_1 = average temperature on the gas side
ϑ_1 = wall temperature on the gas side
t_2 = temperature of cooling agent
ϑ_2 = wall temperature on cooling agent side
a_1 = heat transfer coefficient from gas to wall
a_2 = heat transfer coefficient from wall to cooling agent
k = over-all heat conduction coefficient
λ = heat coefficient through wall
δ = wall thickness (m)
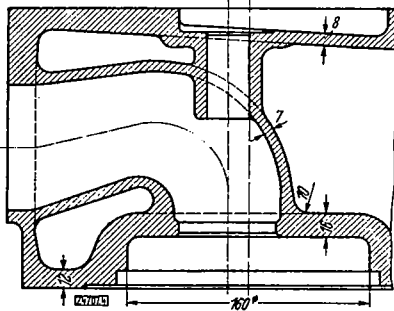
Figure 4.- Cylinder head of a Diesel engine.

Prechamber insert

Cold water
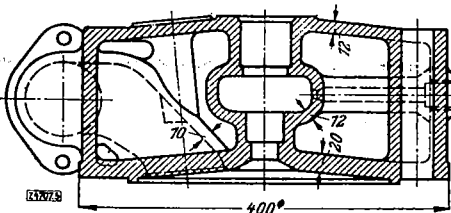
Figure 5.- Cylinder head of a prechamber Diesel engine.
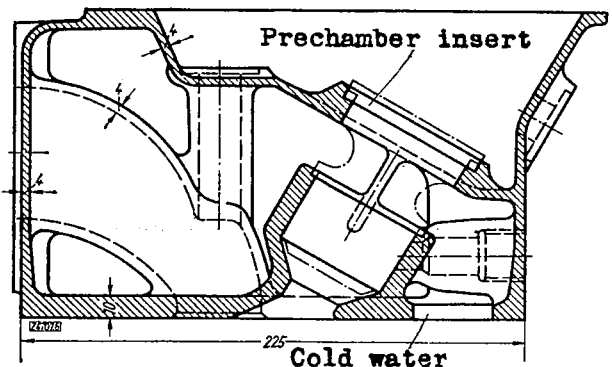
Figure 6.- Cylinder head of a prechamber Diesel engine.
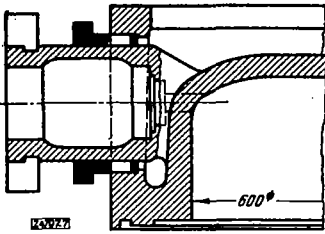
Figure 7.- Safety or starting valve opening in a large Diesel engine.

Figure 8.- Forged iron, water-cooled working piston of a large Diesel engine.
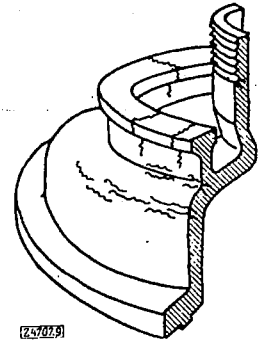
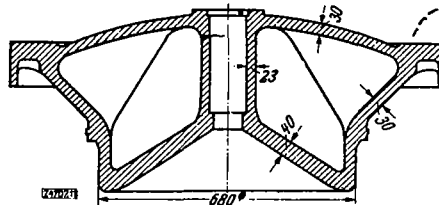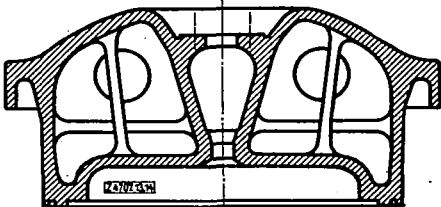Figure 9.- Positions at which cracking takes place in stiffened cylinder head shapes.
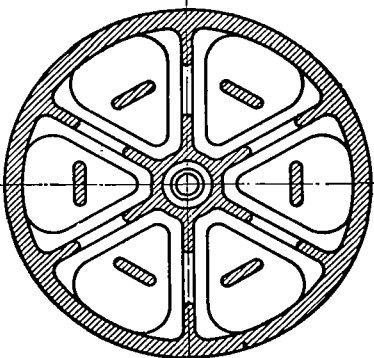
Figure 10.- Rib cross sections.
a, unfavorable section
b, improved section.

Figure 11.- Valve opening in cylinder head of large engine split by heat stresses.

Figure 12.- Cylinder head of a large Diesel engine.

Figures 13,14.- Experimental design of a cylinder cover for large Diesel engines. A thin yielding bottom plate fixed to a strongly stiffened cold cover, radial by yielding supporting ribs.
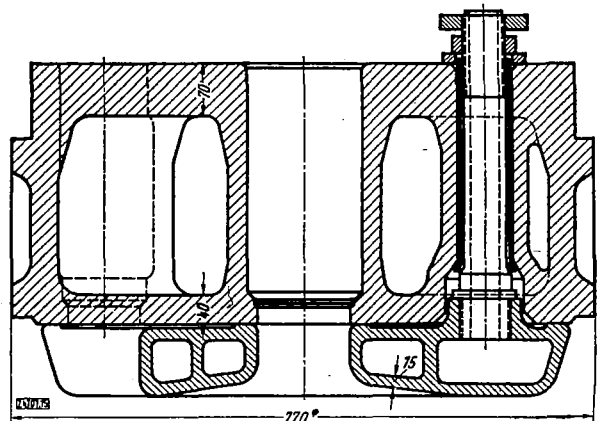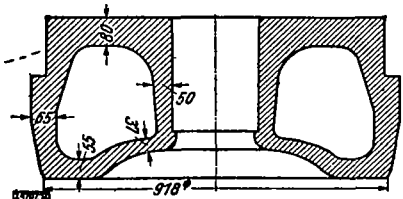
Figure 15.- Cooled, freely suspended added piece in combustion chamber of a large Diesel engine.
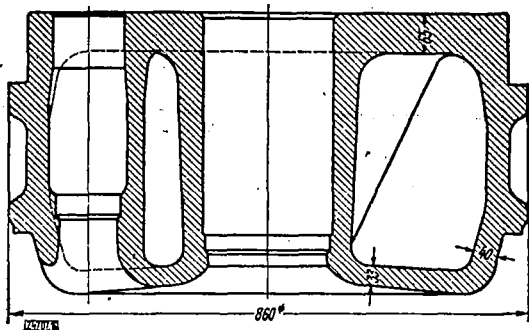
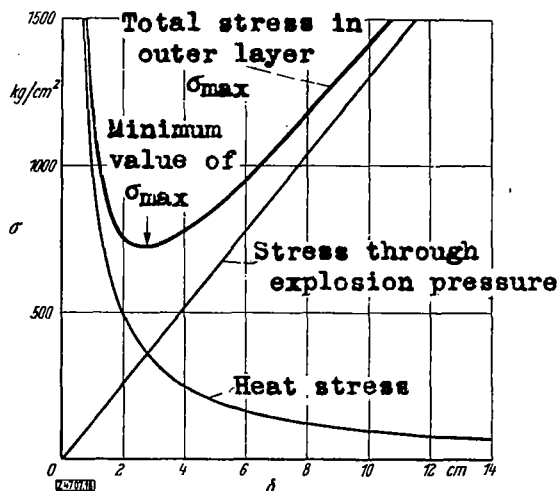Figure 16.- Stiff cylinder head with yielding cover plate.

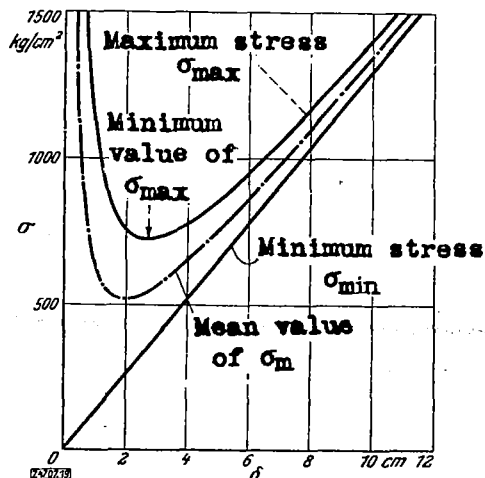Figure 18.- Stress of a cylinder liner as a function of the wall thickness δ

Figure 17.- Liner of a spark ignition engine.

Figure 19.- Fluctuating stress plotted against mean stress σ_m.
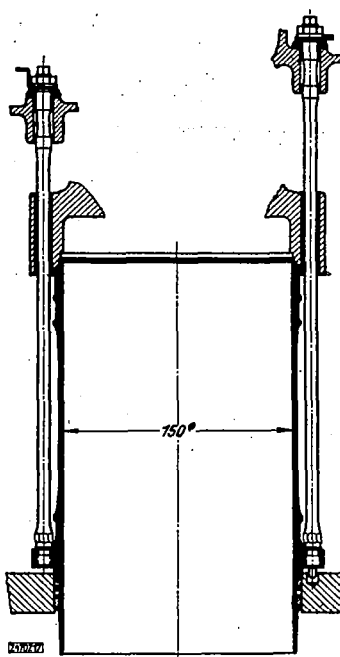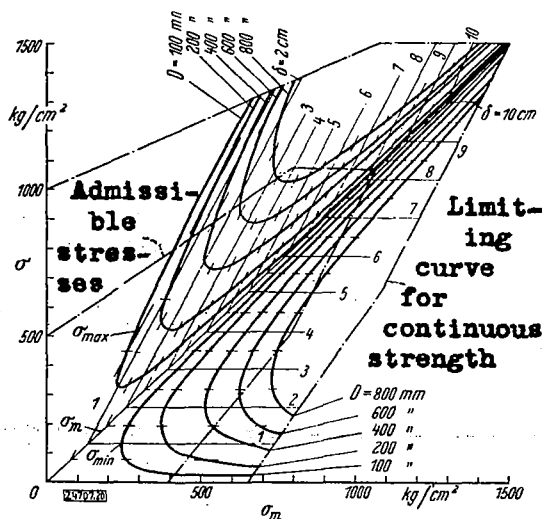

Figure 20.- Maximum stress, mean stress and minimum stress plotted against σ_m for different cylinder diameters.